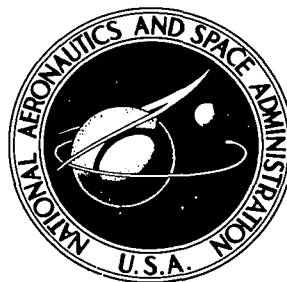


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PREDICTION OF LOCAL
AND INTEGRATED HEAT TRANSFER
IN NOZZLES USING AN INTEGRAL
TURBULENT BOUNDARY LAYER METHOD

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16. Abstract An empirical modification of an existing integral energy turbulent boundary layer method is proposed in order to improve the estimates of local heat transfer in converging-diverging nozzles and consequently, provide better assessments of the total or integrated heat transfer. The method involves the use of a modified momentum-heat analogy which includes an acceleration term comprising the nozzle geometry and free stream velocity. The original and modified theories are applied to heat transfer data from previous studies which used heated air in 30°-15°, 45°-15°, and 60°-15° water-cooled nozzles.					
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PREDICTION OF LOCAL AND INTEGRATED HEAT TRANSFER IN NOZZLES USING AN INTEGRAL TURBULENT BOUNDARY LAYER METHOD

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SUMMARY

An integral turbulent boundary layer theory was applied to heat transfer data from previous heated air studies in $30^\circ\text{-}15^\circ$, $45^\circ\text{-}15^\circ$, and $60^\circ\text{-}15^\circ$ water-cooled nozzles. The data for the $30^\circ\text{-}15^\circ$ and $60^\circ\text{-}15^\circ$ nozzles were obtained at a maximum stagnation pressure and temperature of 207 N/cm^2 absolute and 539 K , respectively. The $45^\circ\text{-}15^\circ$ nozzle data was obtained from a different facility which provided a stagnation pressure and temperature of 172 N/cm^2 , absolute and 811 K , respectively.

The results of this study indicated that a general improvement in estimates of local heat transfer and integrated or total heat transfer in the nozzle could be obtained by altering the original theory (energy method). Specifically, the original auxiliary equation in the energy method was modified to include an acceleration term involving the nozzle geometry and free stream velocity.

INTRODUCTION

The convective heat transfer associated with a turbulent boundary layer in highly accelerated flows has been the subject of numerous studies in the last decade. The previously reported work of references 1 to 4 disclosed that (1) changes in upstream boundary layer momentum history (resulting from different length uncooled inlets) were not important in establishing the level of heat transfer in a nozzle, and (2) the energy thickness Reynolds number in the integral boundary layer analysis of reference 5 provided a good criterion for estimating throat heat transfer. However, estimates of the heat transfer in the nozzle entrance region, based on the latter integral boundary layer theory as well as the differential (similar-solution) analysis of reference 6, were often much lower than the measured values. In current designs of extremely high performance nozzles, improved predictions of local heat flux in regions other than the nozzle throat are

required in order to more accurately determine the overall nozzle performance. The performance can depend, in part, on an assessment of the integrated or total heat loss in the nozzle.

In the present study an empirical modification of the integral boundary layer theory of reference 5 is presented in order to provide a better description of the nozzle heat transfer distribution. The local values of heat flux will be integrated along the nozzle to indicate the differences in predicted total heat flux resulting from the modified theory. The boundary layer analysis is applied to the data of previous nozzle heat transfer studies conducted in a heated air facility at the NASA Lewis Research Center (refs. 1 and 3) and also to data from the heated-air studies of nozzle heat transfer by Back, Massier, and Cuffel (ref. 2).

SYMBOLS

A	nozzle surface area
C_f	skin friction coefficient
C_p	specific heat at constant pressure
D	local diameter
M	Mach number
N	interaction exponent
P	pressure
Pr	Prandtl number
q	heat transfer rate
Q	integrated heat flux, eq. (7)
Q^*	integrated heat flux from nozzle entrance to throat
Q_T	total heat flux in nozzle
R	local radius
Re	Reynolds number
\mathcal{R}	recovery factor
S	distance along wall starting from cone entrance
St	Stanton number
T	temperature
u	velocity

x	axial distance from throat
y	distance from wall
z	axial distance from nozzle entrance
δ	velocity boundary layer thickness
δ^*	displacement thickness
Δ	thermal boundary layer thickness
θ	momentum thickness
ρ	density
φ	energy thickness

Subscripts:

ad	based on adiabatic wall conditions
D	based on local diameter
r	based on reference temperature
s	static condition
t	stagnation condition
w	wall condition
z	axial distance from nozzle entrance
θ	based on momentum thickness
φ	based on energy thickness
∞	condition in the free stream
0	stagnation condition in plenum

Superscripts:

*	geometric throat value
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APPARATUS

The data of references 1 and 3 were obtained at a nominal stagnation temperature of 539 K and pressure of 207 N/cm^2 absolute in the facility at the NASA Lewis Research Center. This facility consisted of a heat exchanger, diffuser, plenum, pipe inlet, and nozzle as shown in figure 1. The heated air was passed into an exhaust system having a nominal pressure of 1.4 N/cm^2 absolute.

The test configurations comprised 16.5-centimeter-diameter uncooled (adiabatic-wall) as well as cooled pipe inlets coupled to water-cooled 30° and 60° half-angle of convergence by 15° half-angle of divergence nozzles (hereinafter referred to as simply the 30° - 15° nozzle, etc.). Details concerning inlet design and dimensions can be obtained in reference 7. The water-cooled 30° - 15° and 60° - 15° nozzles had a nominal throat diameter and throat radius of curvature of 3.8 centimeters. The contraction area ratio given by the ratio of inlet-to-throat cross-section area was approximately 18.8. The 30° - 15° and 60° - 15° nozzle dimensions are presented in tables I and II, respectively.

The data of reference 2 were also obtained in a heated air facility. The test conditions selected for the present report correspond to test number 440 which was conducted at a stagnation temperature of 811 K and pressure of 172 N/cm^2 . The inlet for this test was water cooled and had a diameter of 12.7 centimeters. The water-cooled nozzles had a 45° half-angle of convergence and 15° half-angle of divergence and 4.06-centimeter-diameter throat. The throat radius of curvature to diameter ratio was 0.313. The nominal contraction area ratio for this configuration was 19.8 or approximately the same as that of the 30° - 15° and 60° - 15° nozzles. The dimensions for the nozzle of reference 2 are presented in table III.

INSTRUMENTATION

Since the instrumentation for the 30° - 15° and 60° - 15° nozzles has been described previously (e.g., refs. 1 and 3) only an abbreviated description will be provided herein. The instrumentation pertinent to the present report consisted of wall heat flux meters and wall pressure taps.

The local heat transfer rates, wall temperatures, and wall static pressures in the 30° - 15° and 60° - 15° nozzles were measured at the stations shown in tables I and II, respectively. These measurements were obtained by means of 0.318-centimeter-diameter Inconel heat flux meters and the use of the heat-conduction equation.

Similar measurements were obtained in the 45° - 15° nozzle of reference 2; however, the heat transfer was obtained by a calorimetric technique involving the measurement of water flow rates and corresponding temperature differences in small circumferential passages along the nozzle. The measurements were obtained at the stations shown in table III.

HEAT TRANSFER DATA REDUCTION

The nozzle heat transfer results will be presented in terms of the nondimensional grouping $St_r Pr^{0.7}$ which can be written as

$$St_r Pr^{0.7} = \frac{q}{\rho_r u_\infty C_p (T_{ad} - T_w)} Pr^{0.7} \quad (1)$$

The heat flux q and wall temperature T_w are determined from the temperature measurements on the heat flux meters. The expression for the adiabatic wall temperature T_{ad} is

$$T_{ad} = T_s + \mathcal{R} (T_0 - T_s) \quad (2)$$

where the recovery factor \mathcal{R} for a turbulent boundary layer is equal to $Pr^{1/3}$. The subscript r denotes that properties were evaluated at the Eckert reference temperature T_r (ref. 8) where

$$T_r = T_s + 0.5(T_w - T_s) + 0.22 \mathcal{R} (T_0 - T_s) \quad (3)$$

A Prandtl number of 0.71 was assumed in the previous equations.

EXPERIMENTAL RESULTS

The data used in evaluating the predictive capabilities of the integral boundary layer theory of reference 5 and the modified analysis presented herein were selected in order to emphasize differences in inlet wall cooling, unit Reynolds number, and nozzle convergence conditions. The test cases which were examined are as follows:

- (1) 30°-15° nozzle; $T_0 = 539$ K
 - (a) Adiabatic inlet, $P_0 = 207$ N/cm² (ref. 3)
 - (b) Cooled inlet, $P_0 = 207$ N/cm² (ref. 1)
 - (c) Adiabatic inlet, $P_0 = 110$ N/cm² (ref. 3)
 - (d) Cooled inlet, $P_0 = 110$ N/cm² (ref. 1)
- (2) 60°-15° nozzle; $T_0 = 539$ K
 - (a) Adiabatic inlet, $P_0 = 207$ N/cm² (ref. 3)
 - (b) Cooled inlet, $P_0 = 207$ N/cm² (ref. 1)
- (3) 45°-15° nozzle; $T_0 = 811$ K; cooled inlet, $P_0 = 172$ N/cm² (ref. 2)

The Reynolds numbers resulting from the previous test conditions are considered to be sufficient to produce a turbulent boundary layer in the nozzles in contrast to a "laminarized" boundary layer which was discussed in references 1 and 2.

The experimental values of the Mach number, wall temperature, and heat transfer for the three nozzles are tabulated in tables IV to VI. These basic quantities can be used to convert the heat transfer which is expressed as $St_r Pr^{0.7}$ to other commonly used heat transfer parameters.

The experimental heat transfer $St_r Pr^{0.7}$ is plotted as a function of the axial distance x/D^* in figures 2 to 8. In all cases a minimum in $St_r Pr^{0.7}$ can be noted at or near the geometric throat. This minimum in $St_r Pr^{0.7}$ is characteristic of the flow in converging-diverging nozzles. This effect has been observed to be as low as 50 percent of the value of $St_r Pr^{0.7}$ for unaccelerated flow at the same value of Re_D and fixed free stream conditions (refs. 1 to 4).

The effects of altering the inlet cooling, nozzle convergence angle, and Reynolds number on the nozzle heat transfer were reviewed in reference 7 and therefore, will merely be summarized herein. The effects of inlet cooling can be observed by comparing figures 2 and 3 for the 30° - 15° nozzle or figures 4 and 5 for the 60° - 15° nozzle. In both nozzles, the local heat transfer in tests with the cooled inlet is always lower than values corresponding to tests with the uncooled inlet. The increase in inlet cooling increased the value of the boundary layer energy thickness ϕ at the nozzle entrance and also decreased the nozzle wall temperature (refer to tables IV and V).

The effects of nozzle convergence angle on the heat transfer for the same inlet cooling can best be illustrated by comparing the throat values of $St_r Pr^{0.7}$ in the 30° - 15° and 60° - 15° nozzles since the throat Reynolds numbers were approximately equal. A comparison of the results in figures 2 and 6 or figures 3 and 7 reveals that an increase in convergence angle and, hence, a change in the free stream velocity distribution results in higher values of throat heat transfer. This indicates the importance of the accelerated boundary layer history.

A reduction in stagnation pressure and, consequently, a reduction in Reynolds number tends to increase the value of $St_r Pr^{0.7}$ at a given station in the nozzle. This can be noted by comparing the results for the 30° - 15° nozzle in figures 4 and 5 for a stagnation pressure of 110 N/cm^2 with the corresponding results in figures 2 and 3 for a stagnation pressure of 207 N/cm^2 . The observed trend is predictable on the basis of simple heat transfer relations for unaccelerated flow which indicate that $St_r Pr^{0.7}$ varies approximately as $Re_D^{-0.2}$ (or as $P_0^{-0.2}$ for the data in figs. 2 to 5). Therefore, upon reducing the pressure from 207 to 110 N/cm^2 an increase of about 12 percent in $St_r Pr^{0.7}$ would be expected. The data tend to be quantitatively consistent with this result.

PREDICTED RESULTS

As mentioned previously, the energy method provided good estimates of throat heat transfer; however, in many applications it is also desirable to improve the predictions of heat transfer in the entrance and exit portions of the nozzle. The fundamental criteria set forth by the heat transfer data are as follows: (1) the method must predict the heat transfer at the nozzle entrance (onset of acceleration) and thus reflect the upstream history which, in the present study, was dictated by pipe inlet geometry and cooling, (2) the method must account for the differences in local heat transfer resulting from different levels of acceleration as manifested in the nozzle convergence angle, and (3) the method must account for the effects of Reynolds number on the heat transfer.

With the aforementioned criteria in mind, it is first desirable to review the predictions of heat transfer based on the original formulation of the energy method of reference 5. The method of reference 5 involves the solution of the integral momentum and energy equations and employs Coles friction law, $1/7$ -power profiles for the velocity and temperature difference ratios, and a modified von Kármán form of Reynolds analogy. Details of the method which are pertinent to the present study are given in the appendix.

Application of the Theory of Reference 5

The predicted heat transfer based on the energy method of reference 5 ($N = 0$ in eq. (A1)) is shown for the 30° - 15° and 60° - 15° nozzles in figures 2 to 7 and for the 45° - 15° nozzle (ref. 2) in figure 8. All of the calculations were initialized at the nozzle entrance. The initial values of the boundary layer thicknesses δ and Δ for the tests with cooled inlets (figs. 3, 5, 7, and 8) were obtained by computation from measured values in the pipe inlets using the method of reference 5.

The value of Δ in the nozzle entrance for tests with the uncooled inlet (figs. 2, 4, and 6) was determined by the procedure of reference 3. This procedure entailed varying the value of $(\Delta/\delta)_{z=0}$ until the calculations converged to the same value of throat heat transfer coefficient. Since the thermal boundary layer begins to develop at the nozzle entrance, the value of Δ/δ at this station should be quite small. The value of Δ/δ selected for the present data is the same value that was used in reference 3; namely, $(\Delta/\delta)_{z=0} = 0.01$.

Since, in the energy method, the heat transfer can be calculated independent of the momentum equation, the initial value of momentum thickness does not alter the predicted heat transfer rates. This result is reasonably consistent with the experimental data of reference 9. However, in the predicted results shown in figures 2 to 8, the basic shape of the theoretical distribution differs appreciably from the data. As shown in

reference 3, the energy method provides the best agreement with data in the throat region of the nozzle.

An alternate method of applying the theory is to assume that the interaction exponent N in equation (A1) has a value of 0.25. This is approximately equivalent to incorporating the standard form of the von Kármán momentum heat analogy since the friction coefficient for a turbulent boundary layer usually varies as $Re_\theta^{-0.25}$.

Predictions based on this alternate method are shown for the 30° - 15° nozzle with the uncooled inlet in figure 2. Since the Stanton number in this option is a strong function of Re_θ which is much greater than Re_φ , large differences in predicted heat transfer in the entrance region are apparent. Obviously, predictions of the heat transfer from this option of the theory are not as good as the results based on the energy method (fig. 2).

One final consideration in the method of reference 5 concerns the $1/7$ -power-profile assumption for the temperature and velocity in the boundary layer. Although other power laws have been suggested in the literature, it was shown in reference 9 that the predicted heat transfer resulting from an energy calculation was insensitive to the power law assumption.

Predicted Results Based on a Modification of the Theory of Reference 5

Significant improvements in the predicted heat transfer distribution could not be obtained by using different combinations of the options in the method of reference 5. The predicted heat transfer appears to depend principally on the form of the auxiliary equation which relates the Stanton number to an energy thickness Reynolds number function (eq. (A1)). In order to improve the predictions of heat transfer by the energy method, the limiting case of a pipe flow was considered. In a pipe-cone configuration the heat transfer in the entrance region of the nozzle ($du_\infty/dz \leq 0$) would be dictated in part by the length of pipe inlet preceding the convergent portion of the nozzle. For the Reynolds number levels and entrance conditions of the present nozzles, an expression for the Stanton number which generally improved the agreement with the data in the nozzle entrance region (where the velocity gradient is approximately zero) was determined to be

$$St = 1.6 f(Re_\varphi) \quad (4)$$

where the Reynolds number function is

$$f(\text{Re}_\varphi) = \frac{\frac{C_f(\text{Re}_\varphi)}{2}}{1 - 5 \left[\frac{C_f(\text{Re}_\varphi)}{2} \right]^{1/2} \left[1 - \text{Pr} + \ln \left(\frac{6}{5\text{Pr} + 1} \right) \right]} \quad (5)$$

As the flow accelerates in the convergent portion of the nozzle, the heat transfer is attenuated from the pipe flow level. Upon examining several nondimensional acceleration parameters it was empirically determined that the heat transfer could be correlated in terms of $f(\text{Re}_\varphi)$ and the acceleration parameter $S d(\ln u_\infty R)/dS$, resulting in the following expression for the Stanton number:

$$\text{St} = 1.6f(\text{Re}_\varphi) \left[1.0 - 0.11S \frac{d(\ln u_\infty R)}{dS} \right] \quad (6)$$

The length S is measured along the wall starting at the nozzle entrance. The constant of 0.11 in equation (6) was determined by matching the predicted results to the 30° - 15° nozzle data in figure 2.

The Stanton number in equation (9) was quite sensitive to the acceleration parameter $S d(\ln u_\infty R)/dS$; therefore, it was found desirable to smooth the $\ln u_\infty R$ term before performing the numerical differentiation. Each smooth value of $\ln u_\infty R$ was obtained by evaluating at the corresponding distance z the least square polynomial of degree one relevant to z and the two adjacent points. The numerical differentiation was performed by means of a spline-fit technique.

The results in figures 2 to 8 indicate that, in general, the modified analysis yields a better distribution of heat transfer than the original analysis of reference 5, particularly downstream of the throat. The local heat transfer prediction near the nozzle entrance appears to be greatly improved in the tests with cooled inlets (figs. 3, 5, 7, and 8). No improvement over the reference 5 method is apparent for the cases in which uncooled inlets were used (figs. 2, 4, and 6); however, in the latter cases, the shape of the distribution in the subsonic portion of the nozzle is generally more consistent with the data when based on the modified analysis.

In the majority of cases an improvement in the prediction of the throat heat transfer is apparent with the use of the modified analysis. The throat data in figure 4, corresponding to the 30° - 15° nozzle uncooled inlet configuration operating at $P_0 = 110 \text{ N/cm}^2$, are in better agreement with the results based on the method of reference 5.

Integrated Heat Transfer in the Nozzle

The total heat flux in the nozzle can be determined by integrating the local values of q over the nozzle surface area as shown by the following equation:

$$Q = \int^A q \, dA \quad (7)$$

Equation (7) can be expressed in terms of the $St_r Pr^{0.7}$ grouping and the axial distance from the inlet z as

$$Q(z) = \frac{2\pi}{Pr^{0.7}} \int_0^z St_r Pr^{0.7} \rho_r u_\infty R C_p (T_{ad} - T_w) \left[1.0 + \left(\frac{dR}{dz} \right)^2 \right]^{1/2} dz \quad (8)$$

where the Prandtl number Pr is assumed to be constant.

Values of the total heat transfer from the inlet to the throat Q^* and from the inlet to the downstream station Q_T are tabulated in table VII for the seven nozzle flows considered in this study.

The results in table VII indicate that the modified analysis generally overpredicts the integrated heat transfer from the nozzle entrance to the throat Q^* , whereas the method of reference 5 tended to underpredict Q^* . The differences in predicted values of Q^* are insufficient to preclude one method in favor of the other. This result is especially interesting in the cases which had cooled inlets coupled to the nozzles, since, as shown previously, the modified analysis provided better agreement with the entrance values of $St_r Pr^{0.7}$ and approximately equal or better predictions of throat heat transfer. The underprediction of entrance heat transfer in these cases is offset by the overprediction in the throat of the nozzle so that the method of reference 5 provides good estimates of the integrated subsonic heat transfer. With the exception of the $45^\circ-15^\circ$ nozzle data the agreement in Q^* was within 15 percent. In the latter data, the method of reference 5 underpredicted Q^* by 24 percent. The maximum overprediction by the modified theory was 18 percent.

The modified theory provided better agreement with experimental values of total heat transfer Q_T (integrated heat transfer from the nozzle entrance to exit station) in all cases except for the $60^\circ-15^\circ$ nozzle and cooled inlet configuration operating at $P_0 = 207 \text{ N/cm}^2$ (fig. 7). In this latter case the modified theory overpredicted the experimental values of Q_T by 16 percent. In all of the other cases the agreement was within 6 per-

cent. The method of reference 5 tended to underestimate the total heat transfer by from 8 to 26 percent.

SUMMARY OF RESULTS

An integral turbulent boundary layer theory was applied to heat transfer data from previous studies with heat air in 30°-15°, 45°-15°, and 60°-15° water-cooled nozzles. In general, improved predictions of local heat transfer and consequently the total or integrated heat transfer in the nozzles were obtained by modifying the momentum-heat analogy (auxiliary equation in the standard energy analysis). It was proposed that the original equation, $St = f(Re_\phi)$, be replaced by the following expression for the Stanton number:

$$St = 1.6 f(Re_\phi) \left[1.0 - 0.11S \frac{d(\ln u_\infty R)}{dS} \right]$$

where St is the Stanton number, Re_ϕ is the Reynolds number based on energy thickness, S is the distance along the wall starting from the cone entrance, u_∞ is the velocity in the free stream, and R is the local radius. The predicted results indicated the following:

1. When the original theory ($St = f(Re_\phi)$) was applied to data in which cooled inlets were coupled to the cooled nozzles, an underprediction of the nozzle entrance and exit heat transfer and overprediction of throat heat transfer resulted. The modified theory generally provided better estimates of local heat transfer throughout the nozzle. Although the original method provided better estimates of heat transfer in the nozzle entrance region for the tests in which uncooled inlets were coupled to the cooled nozzles, the modified theory generally predicted a subsonic nozzle heat transfer distribution more consistent with the data. A pronounced improvement in nozzle exit values of predicted heat transfer was obtained when the modified theory was used.

2. The integrated heat transfer in the subsonic portion of the nozzle was generally greater than experimental values when the modified theory was used and less than experimental values when the original analysis was employed. Although the original theory often underpredicted the local heat transfer in the nozzle entrance region and overpredicted the throat heat transfer, the integrated heat transfer in the subsonic portion of the nozzle was within about 15 percent of the data because of these compensating effects.

A pronounced improvement in the agreement between predicted and experimental total heat transfer in the nozzle was obtained when the modified theory was used. With the exception of one case the total heat transfer in the nozzle was within 6 percent of the

experimental values. The original energy analysis generally provided agreement within 18 percent of the integrated values of experimental heat transfer. Estimates of total heat transfer from the latter method were always less than the experimental values.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, December 13, 1971,
132-15.

APPENDIX - REVIEW OF BOUNDARY LAYER THEORY OF REFERENCE 5

The boundary layer theory of reference 5 involves the simultaneous solution of the integral momentum and energy equations in conjunction with auxiliary relations for the skin friction coefficient, boundary layer velocity and temperature profiles, and momentum-heat analogy. All predictions of heat transfer which are based on the method of reference 5 will incorporate the following relations for these auxiliary conditions.

Skin Friction Coefficient

The skin friction coefficient C_f was determined by assuming that it was equal to the adiabatic value $C_{f,ad}$ obtained when $T_w = T_{ad}$. The value of $C_{f,ad}$ was determined by the method of Coles (see ref. 5 for details).

Interaction Exponent

An interaction exponent N is used to relate the Stanton number for unequal momentum and energy thickness to that for equal thickness by means of a factor $(\varphi/\theta)^N$ as shown in the following equation:

$$St = \frac{\frac{C_f(Re_\varphi)}{2} \left(\frac{\varphi}{\theta}\right)^N}{1 - 5 \left[\frac{C_f(Re_\varphi)}{2} \right]^{1/2} \left[1 - Pr + \ln \left(\frac{6}{5 Pr + 1} \right) \right]} \quad (A1)$$

When $\varphi = \theta$, equation (A1) reduces to the familiar von Kármán form of Reynolds analogy. When $N = 0$, equation (A1) can be solved in conjunction with the integral energy equation for the Stanton number. In the present study the interaction exponent N will have a value of zero (energy calculation) unless otherwise specified.

Power Law

The 1/7-power law assumption in the method of reference 5 which apply to the velocity and temperature difference ratios are

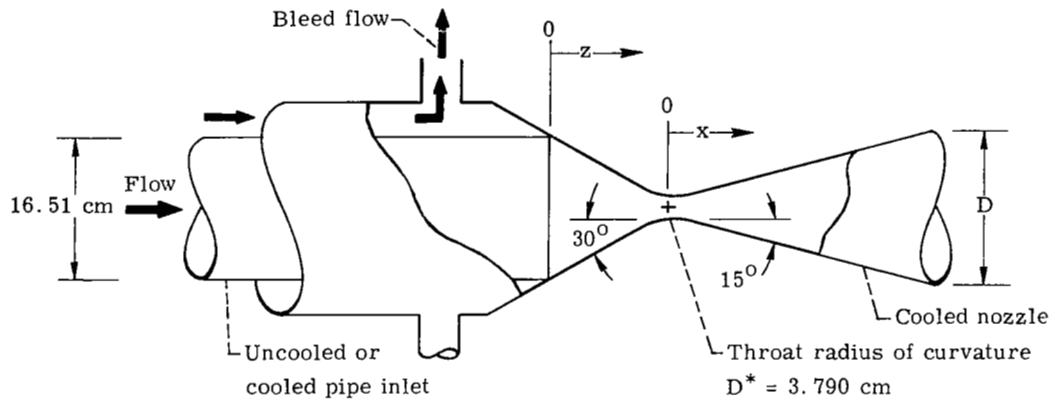
$$\frac{u}{u_{\infty}} = \left(\frac{y}{\delta} \right)^{1/7} \quad (\text{A2})$$

$$\frac{(T_t - T_w)}{(T_0 - T_w)} = \left(\frac{y}{\Delta} \right)^{1/7} \quad (\text{A3})$$

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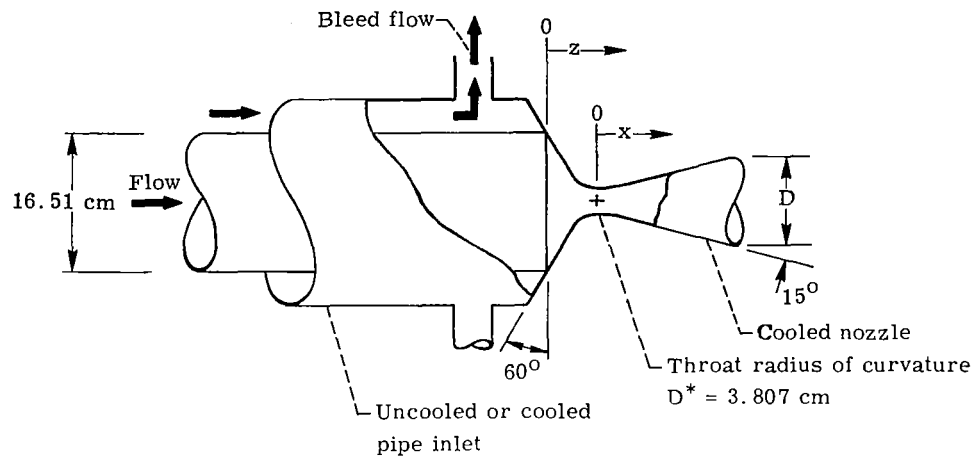
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TABLE I. - INSTRUMENTATION SITES FOR 30°-15° NOZZLE



Station	Axial distance from nozzle entrance, z, cm	Axial distance from nozzle throat, x, cm	Diameter, D, cm	Nondimensional distance from nozzle throat, x/D*
Entrance	0	-12.032	16.510	-3.175
1	.564	-11.468	15.875	-3.026
2	3.104	-8.928	12.934	-2.356
3	5.652	-6.380	9.992	-1.684
4	6.551	-5.481	8.961	-1.446
5	7.430	-4.602	7.945	-1.214
6	8.324	-3.708	6.914	-.979
7	9.213	-2.819	5.883	-.744
8	10.475	-1.557	4.470	-.411
9	11.587	-.445	3.835	-.117
10	12.032	0	3.790	0
11	12.362	.330	3.815	.087
12	12.680	.648	3.912	.171
13	13.028	.996	4.074	.263
14	13.642	1.610	4.399	.425
15	15.133	3.101	5.187	.818
16	18.981	6.949	7.259	1.834
17	25.921	13.889	10.978	3.665

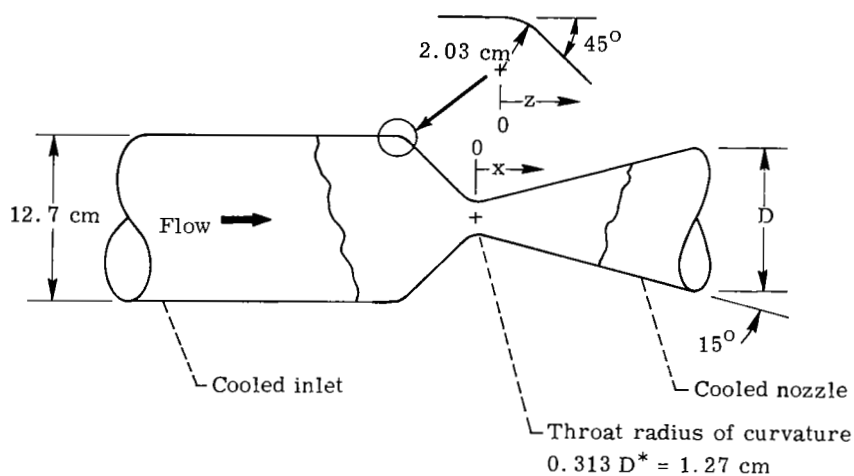
TABLE II. - INSTRUMENTATION SITES FOR 60°-15° NOZZLE



Station	Axial distance from nozzle entrance, z, cm	Axial distance from nozzle throat, x, cm	Diameter, D, cm	Nondimensional distance from nozzle throat, x/D^*
Entrance	0	-5.852	16.510	-1.537
1	.556	-5.296	14.580	-1.391
2	1.402	-4.450	11.643	-1.169
3	1.699	-4.153	10.612	-1.091
4	2.004	-3.848	9.555	-1.011
5	2.789	-3.063	6.939	-.805
6	3.261	-2.591	5.865	-.680
7	4.376	-1.476	4.409	-.388
8	5.481	-.371	3.851	-.097
9	5.852	0	3.807	0
10	6.233	.381	3.863	.100
11	6.556	.704	3.975	.185
12	6.868	1.016	4.122	.267
13	7.432	1.580	4.422	.415
14	8.923	3.071	5.215	.807
15	11.288	5.436	6.464	1.428

TABLE III. - INSTRUMENTATION SITES FOR

45°-15° NOZZLE (REF. 2)



Station	Axial distance from nozzle entrance, z, cm	Axial distance from nozzle throat, x, cm	Diameter, D, cm	Nondimensional distance from nozzle throat, x/D^*
Entrance	0	-5.687	12.715	-1.399
1	.320	-5.367	12.620	-1.321
2	.955	-4.732	12.216	-1.164
3	1.468	-4.219	11.400	-1.038
4	1.961	-3.726	10.441	-.917
5	2.428	-3.259	9.523	-.802
6	2.893	-2.794	8.585	-.688
7	3.358	-2.329	7.654	-.573
8	3.823	-1.864	6.742	-.459
9	4.247	-1.440	5.873	-.354
10	4.625	-1.062	5.119	-.261
11	5.083	-.604	4.365	-.149
12	5.682	-.005	4.064	-.001
13	6.368	.681	4.320	.168
14	7.155	1.468	4.760	.361
15	7.973	2.286	5.195	.563
16	8.832	3.145	5.662	.774
17	10.145	4.458	6.377	1.097
18	11.872	6.185	7.307	1.522
19	13.584	7.897	8.229	1.943
20	15.281	9.594	9.149	2.361

TABLE IV. - TABULATED HEAT TRANSFER DATA FOR 30°-15° NOZZLE

[Throat diameter $D^* = 3.790$ cm.]

Station	Nondimensional distance from nozzle throat, x/D^*	Mach number, M	Uncooled inlet (ref. 3)		Cooled inlet (ref. 1)	
			Stagnation pressure $P_0 = 206.5 \text{ N/cm}^2$	Stagnation temperature $T_0 = 538.8 \text{ K}$	Wall tem- perature, T_w	Heat transfer, $St_{Pr}^{0.7}$
Entrance	1	-3.175	0.02778	389.8	388.9	0.264×10 ⁻²
	2	-2.356	.04678	395.3	395.3	.180
	3	-1.684	.07894	423.9	423.9	.174
	4	-1.446	.09841	433.3	433.3	.161
	5	-1.214	.1266	440.3	440.3	.146
	6	-.979	.1684	447.9	447.9	.125
	7	-.744	.2347	457.8	457.8	.119
	8	-.411	.5055	470.5	470.5	.819×10 ⁻³
	9	-.117	.9281	465.3	465.3	.673
	10	0	1.1144	461.6	461.6	.655
	11	.087	1.2751	457.9	457.9	.737
	12	.171	1.4846	454.7	454.7	.803
	13	.263	1.6811	449.4	449.4	.923
	14	.425	1.7706	443.8	443.8	.880
	15	.818	2.0818	431.3	431.3	.100×10 ⁻²
	16	1.834	2.7889	403.2	403.2	.153
	17	3.665	3.7392	345.0	345.0	.158
	1	-3.026	.02778	363.6	363.9	0.305×10 ⁻²
	2	-2.356	.04678	361.4	361.4	.177
	3	-1.684	.07894	378.7	378.7	.145
	4	-1.446	.09841	391.4	391.4	.147
	5	-1.214	.1266	403.4	403.4	.144
	6	-.979	.1684	415.0	415.0	.135
	7	-.744	.2347	428.3	428.3	.132
	8	-.411	.5055	447.6	447.6	.104
	9	-.117	.9281	448.2	448.2	.887×10 ⁻³
	10	0	1.1144	444.9	444.9	.911
	11	.087	1.2751	442.2	442.2	.967
	12	.171	1.4846	432.1	432.1	.940
	13	.263	1.6811	425.1	425.1	.105×10 ⁻²
	14	.425	1.7706	419.1	419.1	.101
	15	.818	2.0818	406.0	406.0	.114
	16	1.834	2.7889	370.3	370.3	.147
	17	3.665	3.7392	327.1	327.1	.167
	1	-3.175	0.02778	374.4	374.4	0.172×10 ⁻²
	2	-2.356	.04678	389.3	389.3	.138
	3	-1.684	.07894	419.6	419.6	.138
	4	-1.446	.09841	423.8	423.8	.125
	5	-1.214	.1266	427.7	427.7	.106
	6	-.979	.1684	434.6	434.6	.915×10 ⁻³
	7	-.744	.2347	444.4	444.4	.868
	8	-.411	.5055	456.9	456.9	.573
	9	-.117	.9281	457.3	457.3	.527
	10	0	1.1144	457.7	457.7	.542
	11	.087	1.2751	445.9	445.9	.464
	12	.171	1.4846	441.3	441.3	.505
	13	.263	1.6811	436.7	436.7	.608
	14	.425	1.7706	434.5	434.5	.627
	15	.818	2.0818	424.8	424.8	.770
	16	1.834	2.7889	397.8	397.8	.117×10 ⁻²
	17	3.665	3.7392	350.5	350.5	.149
	1	-3.026	.02778	365.9	365.9	0.203×10 ⁻²
	2	-2.356	.04678	367.9	367.9	.136
	3	-1.684	.07894	387.6	387.6	.118
	4	-1.446	.09841	396.2	396.2	.119
	5	-1.214	.1266	405.4	405.4	.112
	6	-.979	.1684	415.1	415.1	.102
	7	-.744	.2347	425.2	425.2	.992×10 ⁻³
	8	-.411	.5055	441.7	441.7	.696
	9	-.117	.9281	443.3	443.3	.642
	10	0	1.1144	440.2	440.2	.629
	11	.087	1.2751	434.2	434.2	.628
	12	.171	1.4846	429.4	429.4	.677
	13	.263	1.6811	423.2	423.2	.756
	14	.425	1.7706	419.8	419.8	.760
	15	.818	2.0818	408.6	408.6	.894
	16	1.834	2.7889	379.2	379.2	.126×10 ⁻²
	17	3.665	3.7392	341.3	341.3	.156

TABLE V. - TABULATED HEAT TRANSFER DATA FOR 60°-15° NOZZLE

[Throat diameter $D^* = 3.807$ cm.]

Station	Nondimensional distance from nozzle throat, x/D^*	Mach number, M	Uncooled inlet (ref. 3)		Cooled inlet (ref. 1)	
			Stagnation pressure $P_0 = 207.0 \text{ N/cm}^2$; Stagnation temperature $T_0 = 537.7 \text{ K}$		Stagnation pressure $P_0 = 207.1 \text{ N/cm}^2$; Stagnation temperature $T_0 = 539.6 \text{ K}$	
			Wall temperature, T_w , K	Heat transfer, $St_{Pr}^{0.7}$	Wall temperature, T_w , K	Heat transfer, $St_{Pr}^{0.7}$
Entrance	-1.537	0.01800	383.9	-----	385.6	-----
1	-1.391	.02300	384.4	0.340×10^{-2}	388.8	0.233×10^{-2}
2	-1.169	.03646	387.9	.234	396.3	.173
3	-1.091	.04989	392.2	.190	400.3	.142
4	-1.011	.06651	402.2	.166	406.3	.122
5	-.805	.1652	440.4	.130	434.1	$.842 \times 10^{-3}$
6	-.680	.2544	460.6	.125	454.2	.810
7	-.388	.5356	473.1	.109	470.9	.749
8	-.097	.9366	476.8	$.990 \times 10^{-3}$	469.6	.601
9	0	1.1004	474.9	$.102 \times 10^{-2}$	467.7	.605
10	.100	1.3471	470.3	.118	461.1	.643
11	.185	1.5289	463.1	.121	457.9	.763
12	.267	1.6384	457.1	.111	453.9	.788
13	.415	1.7907	445.9	.112	445.1	.805
14	.807	2.0944	429.1	.121	431.8	.933
15	1.428	2.5460	406.1	.136	414.5	$.102 \times 10^{-2}$

TABLE VI. - TABULATED HEAT TRANSFER DATA

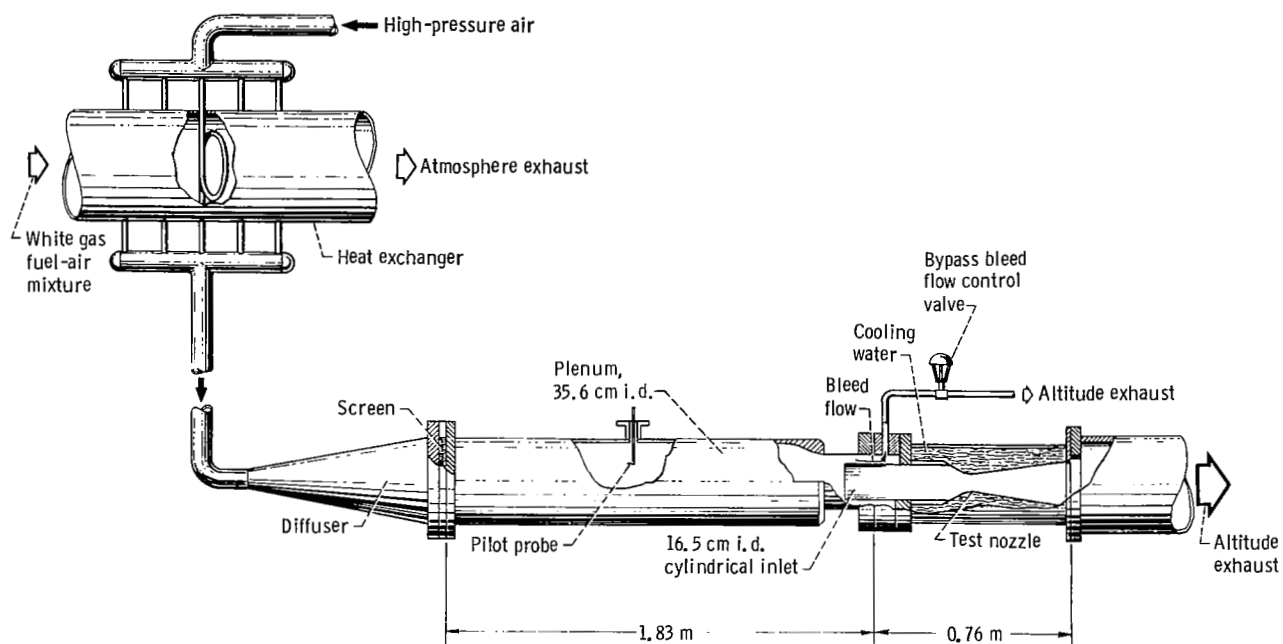
FOR 45° - 15° NOZZLE (REF. 2)[Throat diameter $D^* = 4.064$ cm.]

Station	Nondimensional distance from nozzle throat, x/D^*	Mach number, M	Cooled inlet	
			Stagnation pressure $P_0 = 172.2 \text{ N/cm}^2$; Stagnation temperature $T_0 = 811.1 \text{ K}$	
			Wall temperature, T_w , K	Heat transfer, $St_r Pr^{0.7}$
Entrance	-1.399	0.01800	373.9	0.851×10^{-3}
1	-1.321	.02300	371.7	.764
2	-1.164	.03646	373.9	.982
3	-1.038	.05281	382.2	1.008
4	-.917	.06331	386.1	1.024
5	-.802	.09789	396.7	.978
6	-.688	.1123	408.9	1.070
7	-.573	.1399	417.8	1.104
8	-.459	.1711	426.7	1.075
9	-.354	.1910	428.9	.955
10	-.261	.3147	442.2	.784
11	-.149	.7462	440.6	.577
12	-.001	1.3645	441.1	.686
13	.168	1.7321	437.8	.729
14	.361	1.8058	431.7	.748
15	.563	1.9234	430.0	.771
16	.774	2.0852	422.2	.801
17	1.097	2.3427	416.1	.742
18	1.522	2.6246	404.4	.790
19	1.943	2.9429	396.7	.917
20	2.361	3.1462	397.8	.860

TABLE VII. - EXPERIMENTAL AND CALCULATED TOTAL HEAT TRANSFER IN NOZZLE

Configuration	Figure showing heat transfer	Nominal stagnation temperature, T_0 , K	Nominal stagnation pressure, P_0 , N/cm^2	Total heat transfer from inlet to throat, Q^* , kW			Ratio of calculated to experimental total heat transfer from inlet to throat		Total heat transfer from inlet to downstream station, Q_T , kW			Ratio of calculated to experimental total heat transfer from inlet to downstream station	
				Experiment	Theory (ref. 5)	Modified theory (present analysis)	Theory (ref. 5)	Modified theory (present analysis)	Experiment	Theory (ref. 5)	Modified theory (present analysis)	Theory (ref. 5)	Modified theory (present analysis)
^a 30°-15° nozzle, adiabatic inlet (ref. 3)	2	539	207	5.16	4.82	6.10	0.93	1.18	8.60	7.35	9.01	0.85	1.05
30°-15° nozzle, cooled inlet (ref. 1)	3		207	4.17	3.55	4.68	.85	1.12	6.98	5.69	7.21	.82	1.03
30°-15° nozzle, adiabatic inlet (ref. 3)	4		110	3.77	3.58	4.05	.95	1.07	6.27	5.50	6.19	.88	.99
30°-15° nozzle, cooled inlet (ref. 1)	5		110	2.83	2.53	3.33	.89	1.17	4.83	4.06	5.13	.84	1.06
60°-15° nozzle, adiabatic inlet (ref. 3)	6		207	3.07	2.70	3.05	.88	1.00	4.61	3.98	4.55	.86	.99
60°-15° nozzle, cooled inlet (ref. 1)	7		207	2.22	2.03	2.59	.91	1.17	3.36	3.09	3.90	.92	1.16
45°-15° nozzle, cooled inlet (ref. 2)	8	811	172	6.13	4.68	5.96	.76	.96	12.03	8.90	11.62	.74	.97

^aDenotes configuration used to obtain constants in modified theory.



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Figure 1. - Schematic diagram of NASA Lewis Research Center nozzle heat transfer facility.

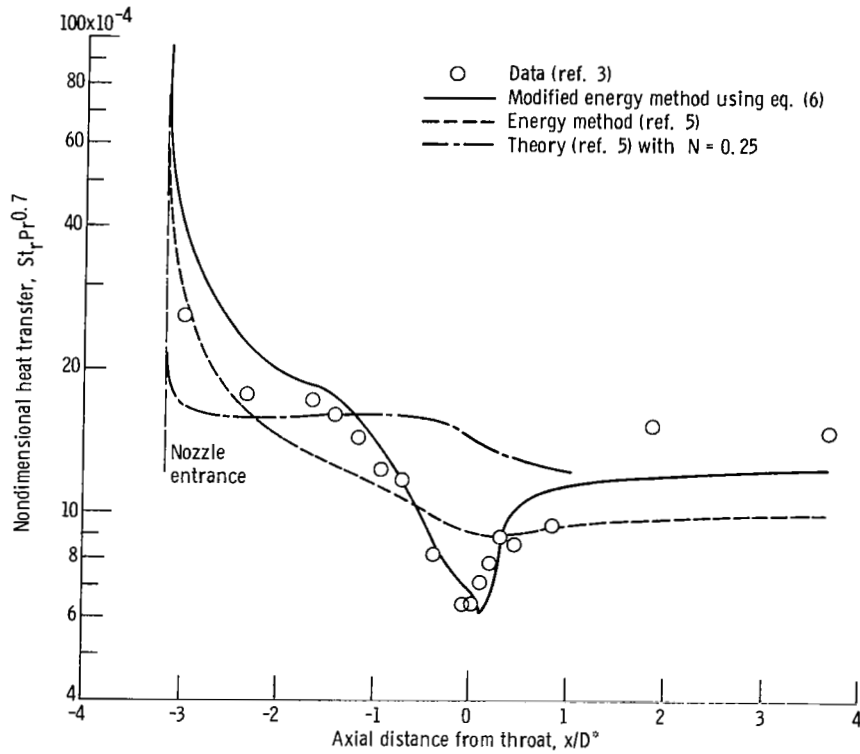


Figure 2. - Comparison of experimental and predicted heat transfer in 30°-15° nozzle operating with uncooled inlet. Stagnation temperature $T_0 = 538.8$ K; stagnation pressure $P_0 = 206.5$ N/cm²; throat diameter $D^{\circ} = 3.790$ centimeters. Deficiency thicknesses at nozzle entrance: energy thickness $\varphi = 0.000452$ centimeter; momentum thickness $\theta = 0.0871$ centimeter; displacement thickness $\delta^* = 0.106$ centimeter.

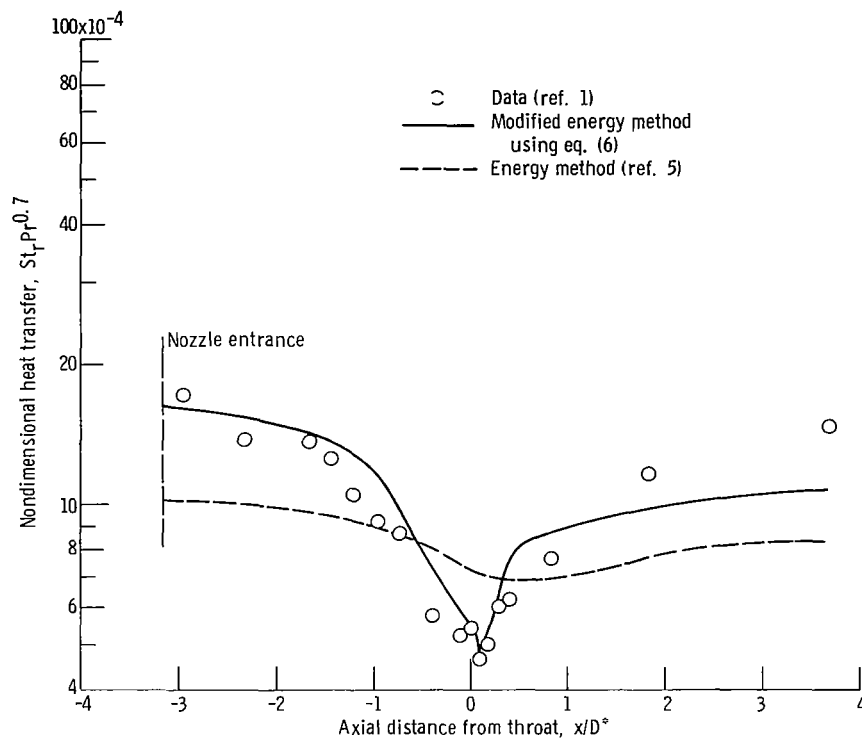


Figure 3. - Comparison of experimental and predicted heat transfer in 30° - 15° nozzle operating with cooled inlet. Stagnation temperature $T_0 = 537.9$ K; stagnation pressure $P_0 = 205.6$ N/cm²; throat diameter $D^* = 3.790$ centimeters. Deficiency thicknesses at nozzle entrance: energy thickness $\varphi = 0.143$ centimeter; momentum thickness $\theta = 0.160$ centimeter; displacement thickness $\delta^* = 0.150$ centimeter.

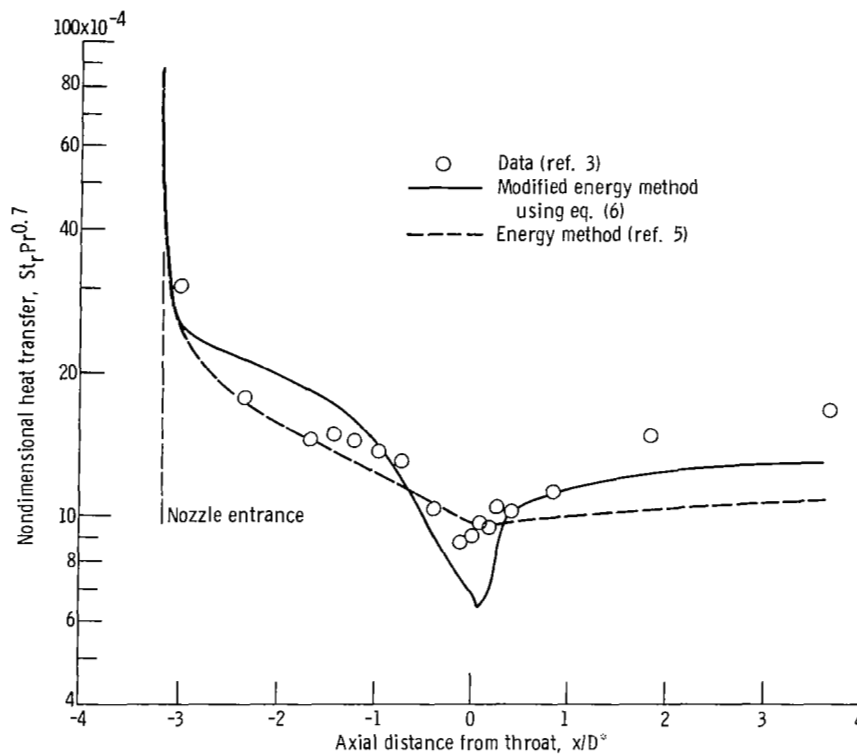


Figure 4. - Comparison of experimental and predicted heat transfer in 30° - 15° nozzle operating with uncooled inlet. Stagnation temperature $T_0 = 539.2$ K; stagnation pressure $P_0 = 108.8$ N/cm 2 ; throat diameter $D^* = 3.790$ centimeters. Deficiency thicknesses at nozzle entrance: energy thickness $\varphi = 0.000511$ centimeter; momentum thickness $\theta = 0.0980$ centimeter; displacement thickness $\delta^* = 0.118$ centimeter.

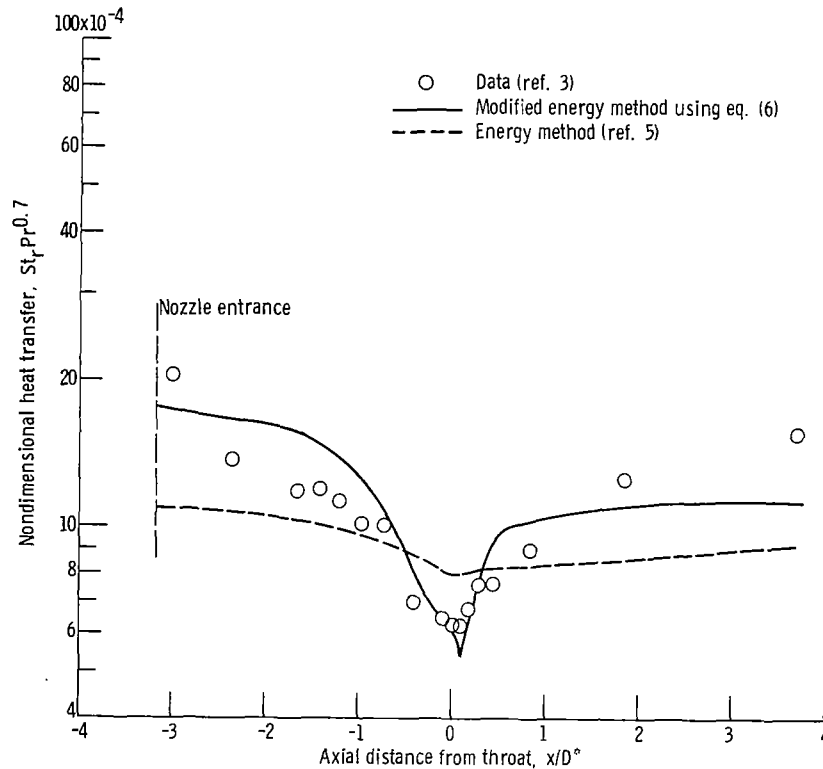


Figure 5. - Comparison of experimental and predicted heat transfer in 30° - 15° nozzle operating with cooled inlet. Stagnation temperature $T_0 = 540.0$ K; stagnation pressure $P_0 = 109.8$ N/cm 2 ; throat diameter $D^* = 3.790$ centimeters. Deficiency thicknesses at nozzle entrance: energy thickness $\varphi = 0.177$ centimeter; momentum thickness $\theta = 0.194$ centimeter; displacement thickness $\delta^* = 0.176$ centimeter.

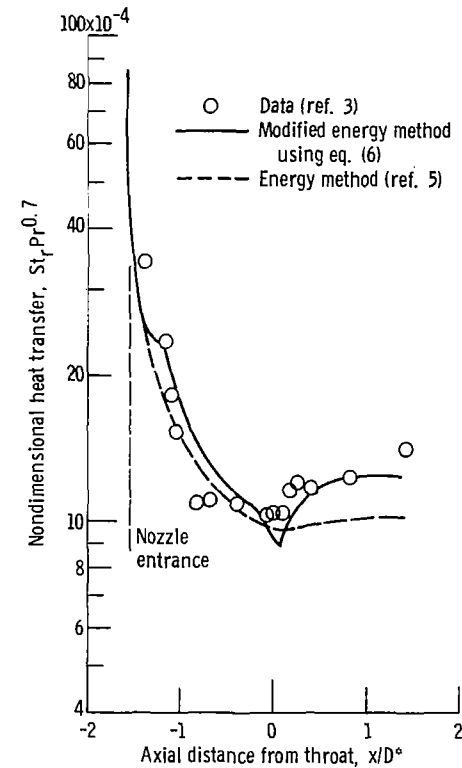


Figure 6. - Comparison of experimental and predicted heat transfer in 60° - 15° nozzle operating with uncooled inlet. Stagnation temperature $T_0 = 537.7$ K; stagnation pressure $P_0 = 207.0$ N/cm 2 ; throat diameter $D^* = 3.807$ centimeters. Deficiency thicknesses at nozzle entrance: energy thickness $\varphi = 0.000439$ centimeter; momentum thickness $\theta = 0.000439$ centimeter; displacement thickness $\delta^* = 0.000406$ centimeter.

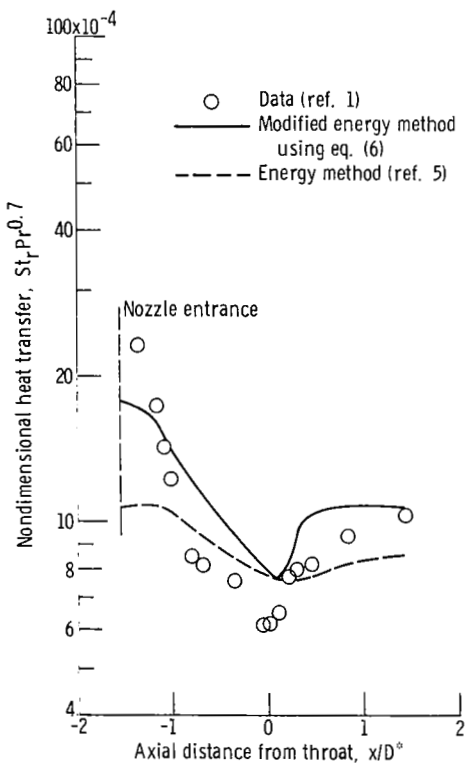


Figure 7. - Comparison of experimental and predicted heat transfer in 60° - 15° nozzle operating with cooled inlet. Stagnation temperature $T_0 = 539.6$ K; stagnation pressure $P_0 = 207.1$ N/cm 2 ; throat diameter $D^* = 3.807$ centimeters. Deficiency thicknesses at nozzle entrance: energy thickness $\varphi = 0.143$ centimeter; momentum thickness $\theta = 0.160$ centimeter; displacement thickness $\delta^* = 0.153$ centimeter.

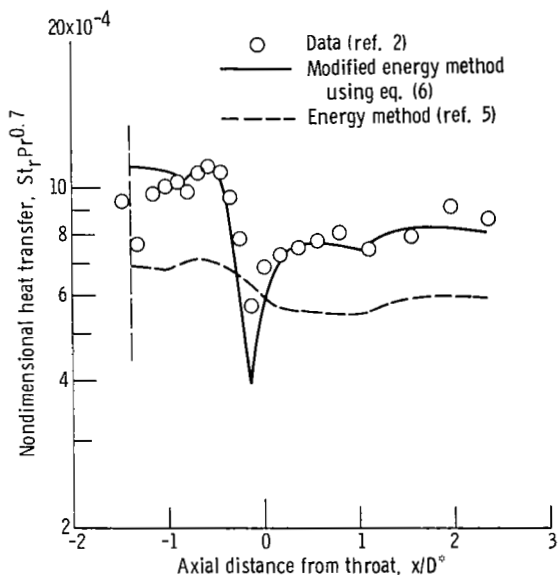


Figure 8. - Comparison of experimental and predicted heat transfer in 45° - 15° nozzle operating with cooled inlet. Stagnation temperature $T_0 = 811.1$ K; stagnation pressure $P_0 = 172.2$ N/cm 2 ; throat diameter $D^* = 4.064$ centimeters. Deficiency thicknesses at nozzle entrance: energy thickness $\varphi = 0.351$ centimeter; momentum thickness $\theta = 0.264$ centimeter; displacement thickness $\delta^* = 0.107$ centimeter.